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(54) I.C. Engine combustion chambers

(57) Fuel is injected into a primary combustion chamber 8 late in the compression stroke and is ignited by a sparking plug 10. A throat 9 is positioned and dimensioned so that air flowing into the chamber 8 rotates therein, the period over which fuel is injected into the chamber 8 corresponding at full load to at most four times the period of a rotation and corresponding at less than full load operation to a shorter period of rotation and preferably to less than the period of one rotation.

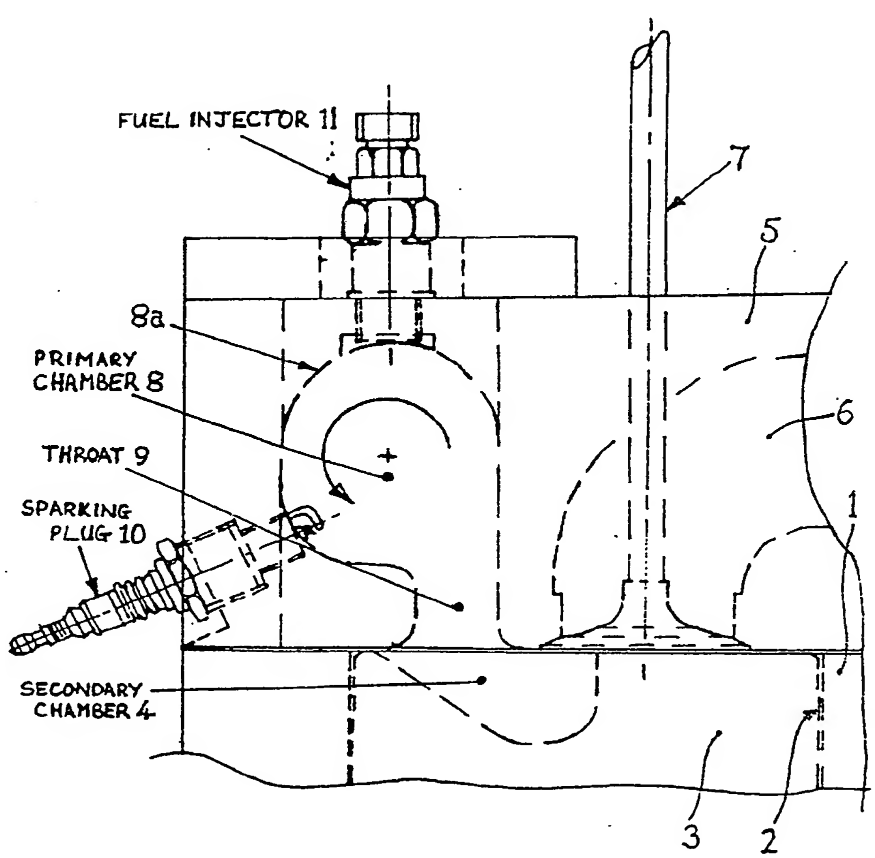

FIG. 1 BASIC CONFIGURATION

The drawings originally filed were informal and the print here reproduced is taken from a later filed formal copy.

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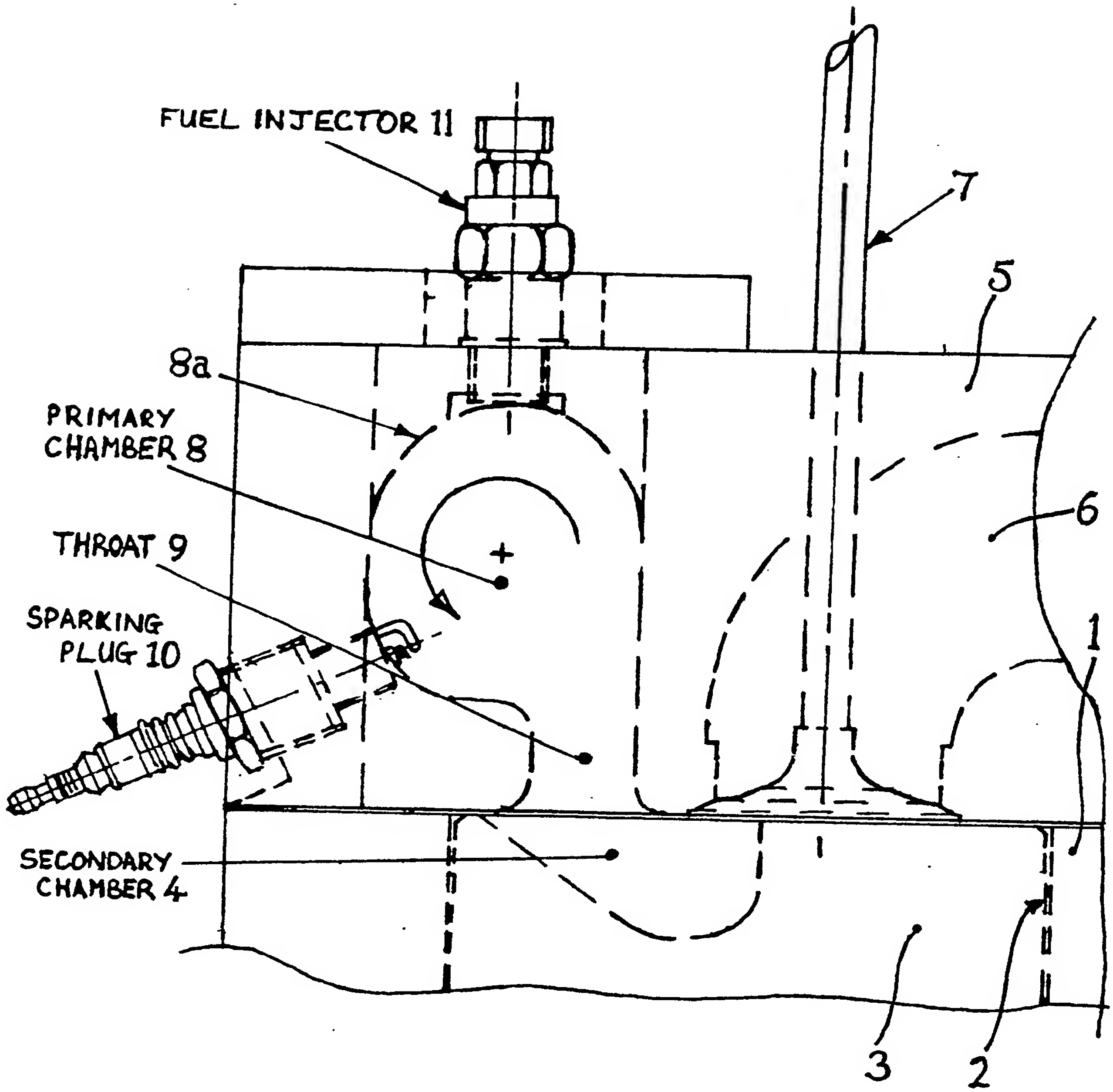


FIG. 1 BASIC CONFIGURATION

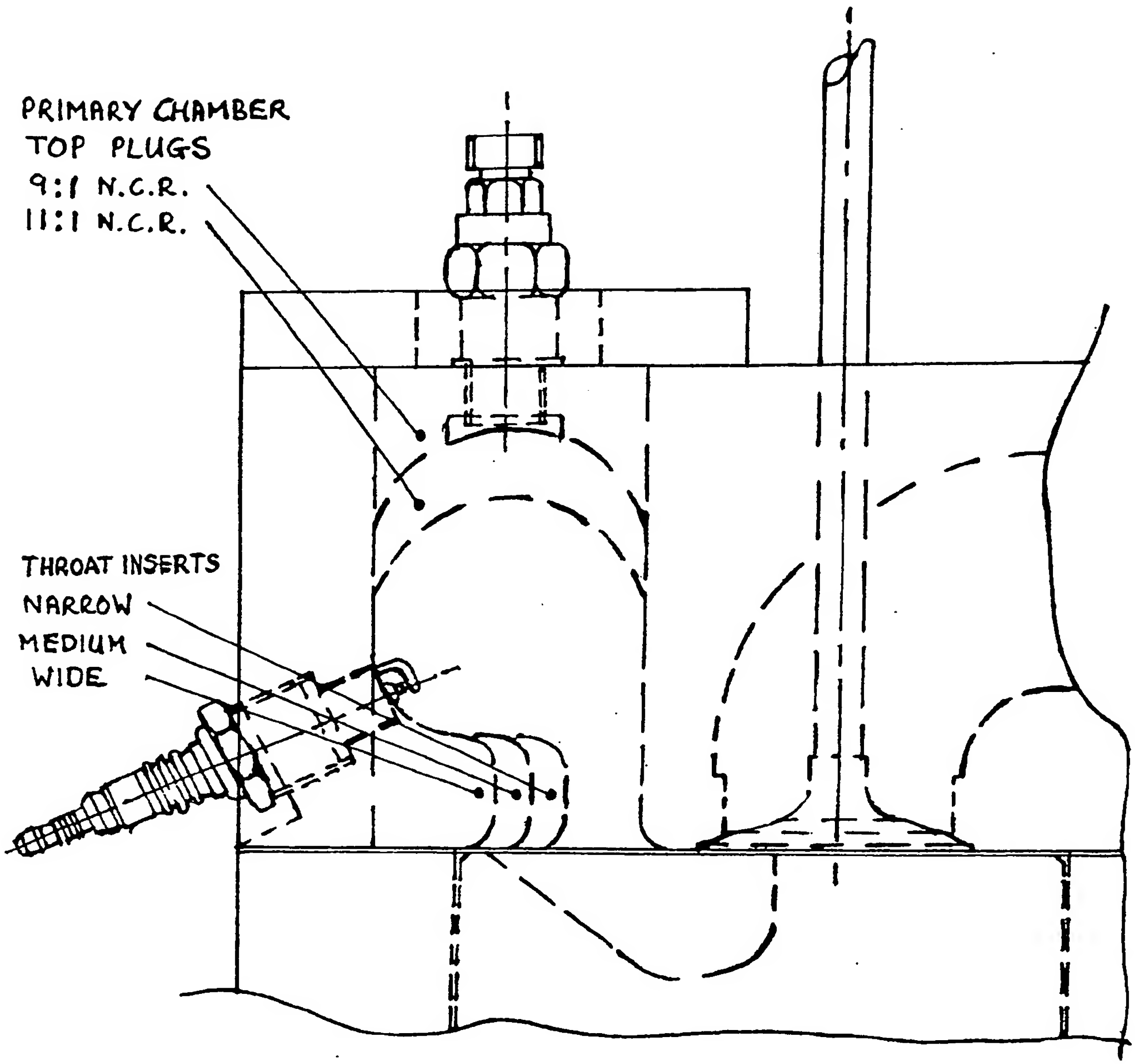
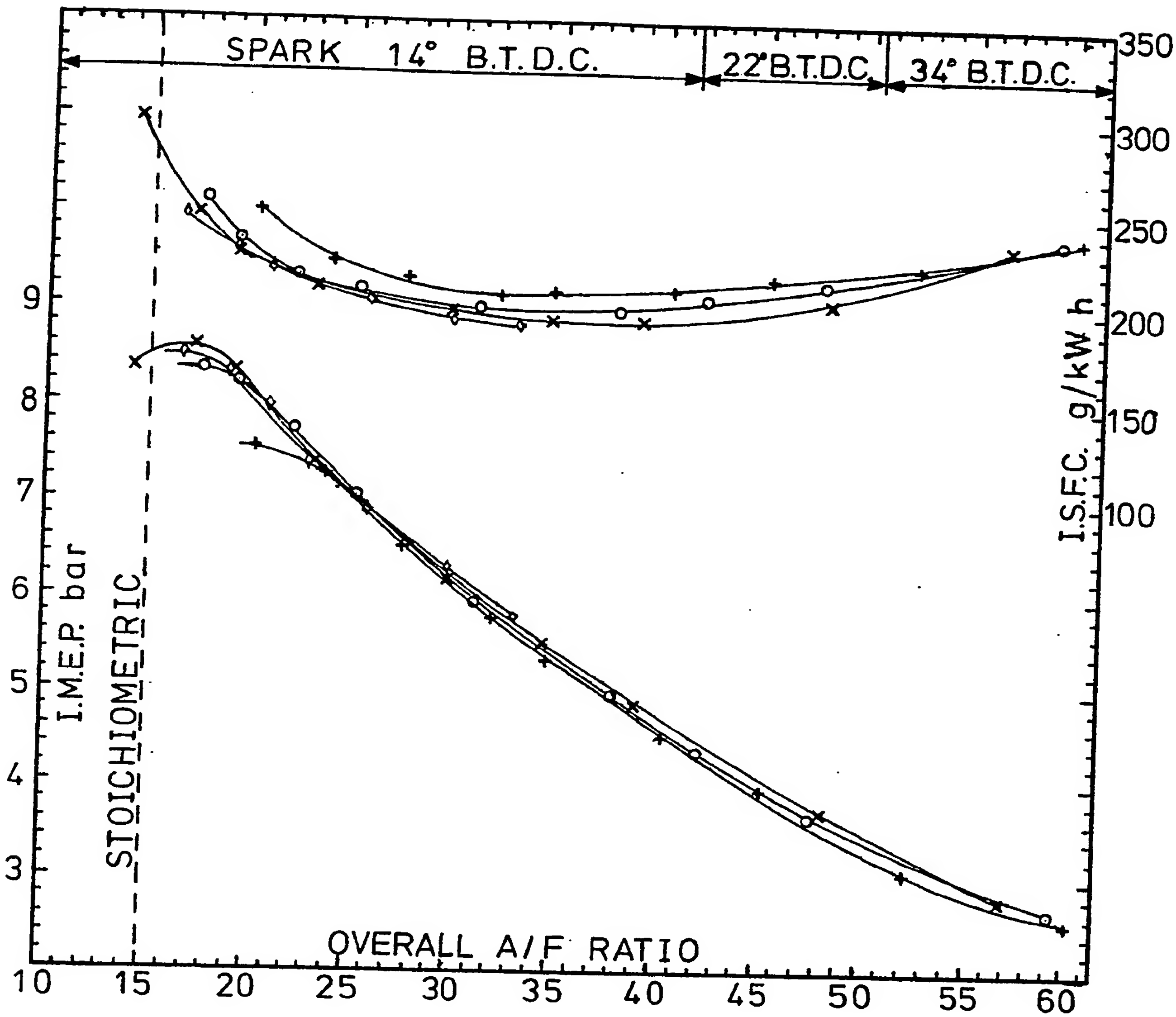


FIG. 2 VARIATIONS IN CHAMBER GEOMETRY



\+ 16·7 rev/s

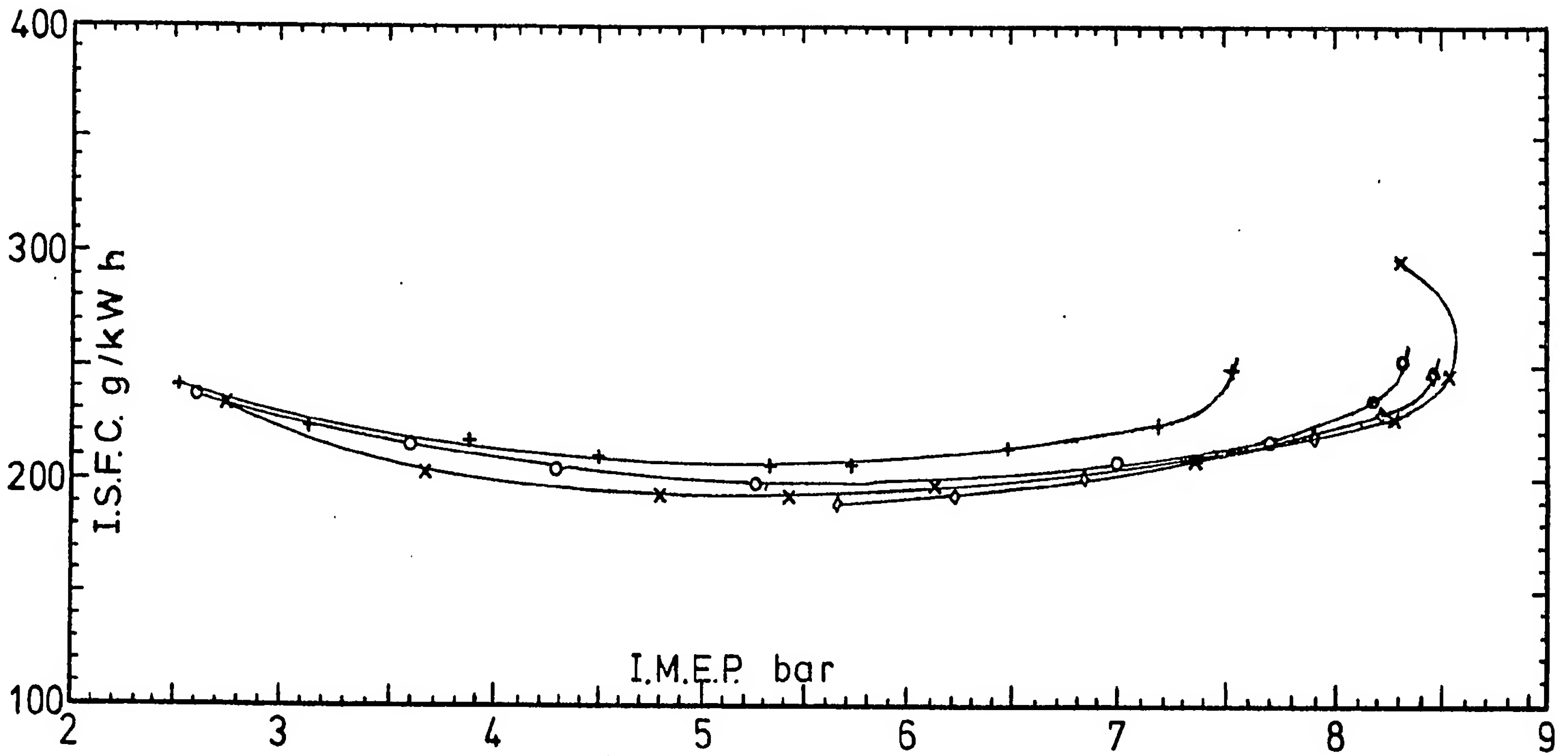
o 18·3 rev/s

x 20·0 rev/s

◇ 22·5 rev/s

INJECTION STATIC TIMING 57° B.T.D.C.

FIG 3 I.M.E.P. & I.S.F.C. v A/F RATIO FOR BASIC CONFIGURATION



+ 16·7 rev/s

o 18·3 rev/s

× 20·0 rev/s

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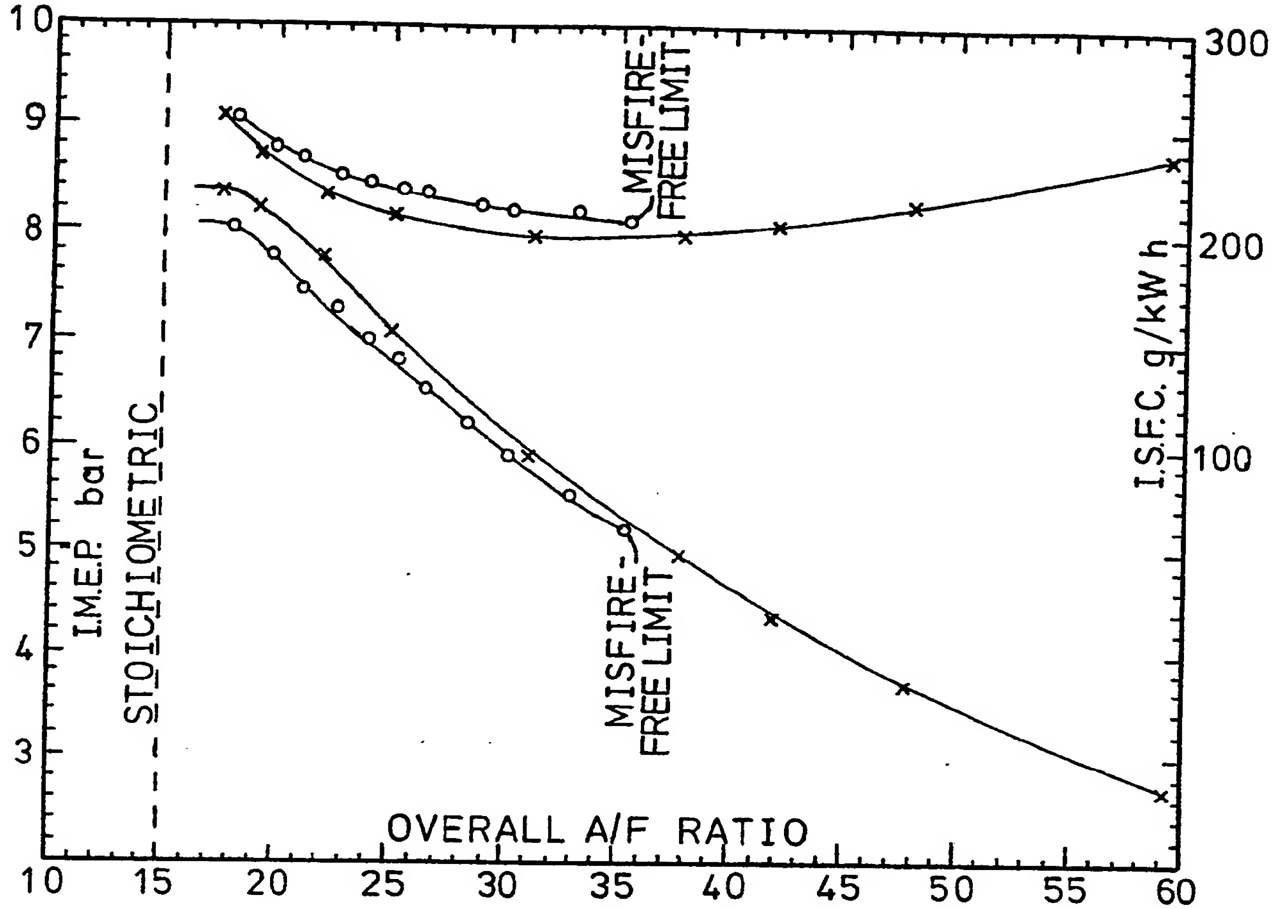
INJECTION STATIC TIMING 57° B.T.D.C.

FIG 4 I.S.F.C. v I.M.E.P. FOR BASIC CONFIGURATION



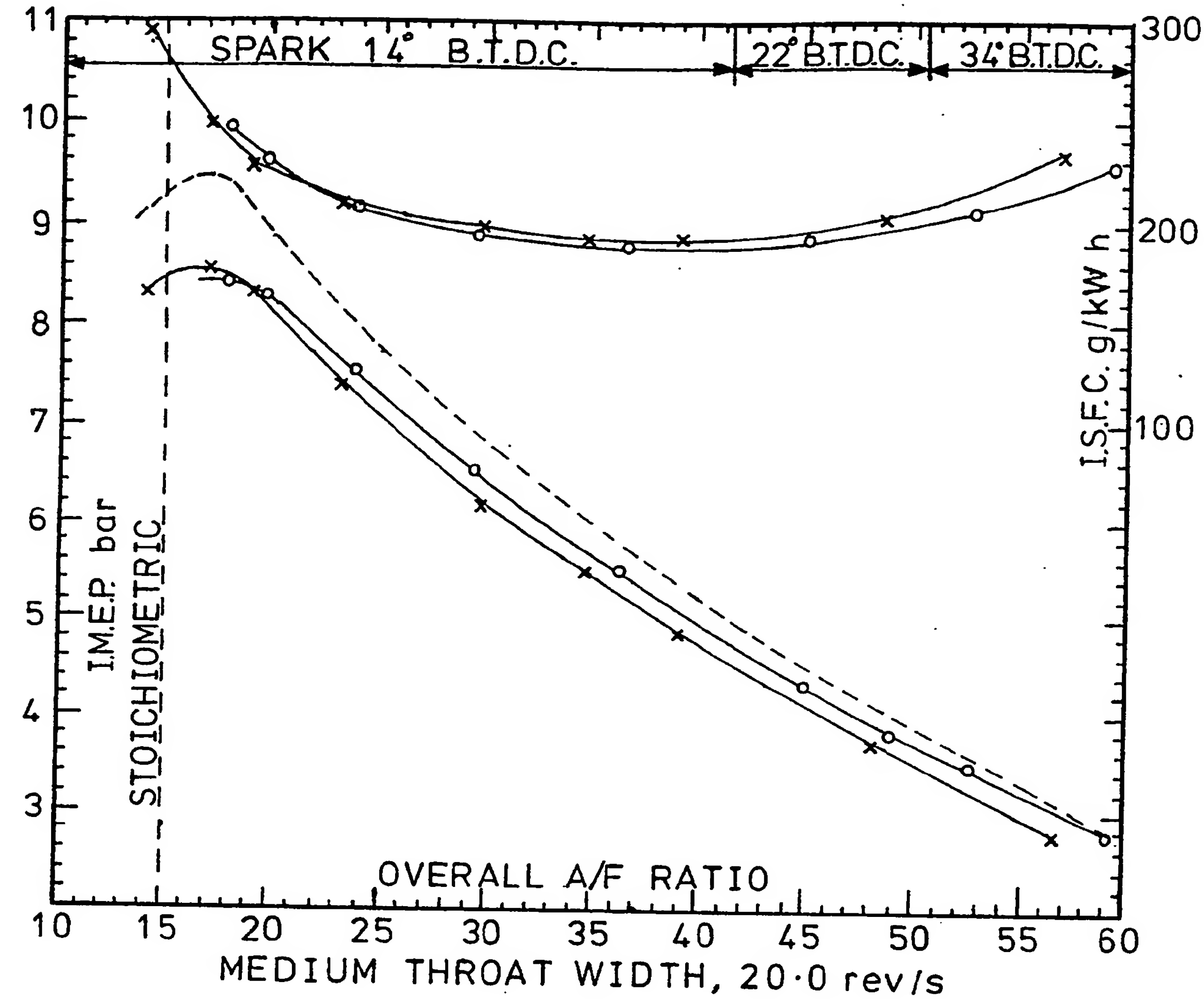
9:1 N.C.R. 18·3 rev/s

× MEDIUM THROAT WIDTH
o NARROW THROAT WIDTH

INJECTION STATIC TIMING: × 57° B.T.D.C.
o 38° B.T.D.C.

SPARK TIMING: × AS IN FIG 3
o 12° B.T.D.C.

FIG 5 COMPARISON OF THROAT WIDTHS



× 9:1 N.C.R.

o 11:1 N.C.R.

INJECTION STATIC TIMING: × 57° B.T.D.C.
o 64° B.T.D.C.

DOTTED LINE SHOWS 9:1 N.C.R. RESULTS SCALED UP TO COMPENSATE FOR INCREASED AIR CYCLE EFFICIENCY AND IMPROVED VOLUMETRIC EFFICIENCY OF 11:1 N.C.R. RESULTS.

FIG 6 COMPARISON OF PRIMARY CHAMBER SIZES

SPECIFICATION

Internal combustion engine

The present invention relates to divided chamber stratified charge engines.

Stratified charge combustion

Stratified charge engines employ a heterogeneous fuel-air mixture, usually by some form of fuel injection, and spark ignition, with unthrottled or reduced throttled induction, and a moderate compression ratio. These engines are thus hybrids with features of both compression ignition and conventional spark ignition engines. The major objectives of such hybridization are to achieve the improved low load fuel economy of the compression ignition engine without sacrificing the lightness, smoothness, quietness and clear exhaust of the spark ignition, carburetted engine.

The principle of stratified charge combustion is that an overall lean fuel-air mixture can be efficiently burned provided a controlled richness around the sparking plug is provided when the spark occurs. Once this portion of the charge has been ignited the flame will spread and engulf the remainder of the mixture, within limits, rich or lean.

Improved efficiency

The main incentive towards this concept for automotive engines is the fuel economy rewards in the part load region where vehicles are mostly driven. Blame for the conventional spark ignition engine's poor fuel consumption at part load is directed at the charge throttle.

At part loads improvements in efficiency as a result of full throttle running are achieved for the following reasons:

1. The fuel-air cycle efficiency is closer to the ideal air cycle efficiency ($\eta=1-1/r^{\gamma-1}$) because the ratio of specific heats is higher for a lean mixture than for a stoichiometric one, there is less variation in the specific heats during the cycle and less dissociation because mean flame temperatures are lower.

2. Because of lower flame temperatures there is less heat loss to the coolant.

3. Air pumping losses are less.

Reduction in engine exhaust pollutants

The major pollutants subject to legislation found in the exhausts of conventional spark ignition engines are unburned hydrocarbons (HCn), carbon monoxide (CO) and oxides of nitrogen (NOx). It is well known that one of the most significant variables affecting exhaust concentrations of these species is the fuel-air ratio supplied to the engine. Fuel-air ratios leaner than stoichiometric are favourable from the standpoint of minimizing unburned hydrocarbons and carbon monoxide, but in conventional spark ignition engines concentrations of oxides of nitrogen are maximised by a slightly leaner than stoichiometric mixture. For this reason the control of unburned hydrocarbon and carbon monoxide emissions is seriously complicated by the added requirement of simultaneous control of oxides of nitrogen.

The kinetics of the reaction are such that nitrogen oxides are formed in significant concentrations only in postflame reactions following the passage of the flame front and once formed remain substantially fixed throughout the expansion and exhaust processes. Some stratified charge engine concepts are designed to exploit this finite time dependence of nitrogen oxides formation in high temperature combustion products. Reactions involving the formation or destruction of oxides of nitrogen generally require temperatures higher than those involving the oxidation of carbon monoxide or hydrocarbons. For combustion system temperatures less than approximately 2200°K, nitrogen is effectively inert relative to the time scale of typical reciprocating internal combustion engine processes. Carbon monoxide and unburned hydrocarbons, however, oxidize readily at such temperatures.

Thus combustion processes can be designed on the basis of the above for the purpose of simultaneously reducing exhaust emissions of hydrocarbons, carbon monoxide and oxides of nitrogen. In principle the process should occur in two stages, beginning with ignition and flame propagation through a well confined fuel-air mixture. Combustion chamber geometry or mixture motion should be such that as the flame propagates through the fuel-air mixture, high temperature combustion products rapidly expand into a region containing relatively low temperature air. As a result the post-combustion gases are quenched to temperatures sufficiently low that little nitrogen oxides formation can occur even though considerable oxygen is present. The resulting temperatures, however are sufficiently high that oxidation of any remaining carbon monoxide and unburned hydrocarbons occurs with the excess air. Therefore the final combustion products exhausted from the engine contain very low concentrations of oxides of nitrogen, hydrocarbons and carbon monoxide.

Multifuel application

Some stratified charge engine designs show the practical, economic value of being able to utilize broad boiling range fuels with no octane or cetane number specification that can be produced from crude oil in high yield and with low processing costs. These engines all have fuel injection occurring late on the compression stroke i.e. commencing shortly before the spark and continuing usually through the beginning of the combustion process. The elimination of combustible end gases trapped for a prolonged interval in remote parts of the chamber considerably reduced octane requirement, whilst the positive ignition by the spark eliminates cetane requirement.

A low-emission predecessor

The combustion chamber concept of

NEWHALL and EL-MESSIRI offers a great advantage from the point of view of reducing emissions, particularly of oxides of nitrogen, through the employment of two distinct phases of combustion. The NEWHALL and EL-MESSIRI engine comprises an unscavenged prechamber or primary chamber occupying 65—85% of the total clearance volume. All fuel is injected into the primary chamber early on the compression stroke of a 4-stroke cycle and at no time is raw fuel allowed to enter the secondary or cylinder chamber. The resulting mixture is ignited by a sparking plug near the throat separating the primary chamber from the secondary chamber. Flame propagation into the mixture away from the throat forces the burning gases rapidly into the secondary chamber where they mix with the colder air. Further oxidation with excess air then occurs at lower temperatures.

However, the NEWHALL and EL-MESSIRI system involves control by a conventional air inlet throttle valve with simultaneous alteration of fuel injection quantity to maintain a fairly uniform and constant, slightly richer than stoichiometric air-fuel ratio in the primary chamber under all load conditions, though the overall air-fuel ratio of the engine is at all times leaner than stoichiometric. An engine built to this system produced very low emission levels but showed relatively poor economy.

A controlled combustion system

The object of the present invention is to provide a new stratified charge engine wherein wide-open throttle operation with a primary chamber occupying say 75% of the total clearance volume can be accomplished by the careful coordination of injection late on the compression stroke and controlled air swirl to selectively distribute fuel in the primary chamber to give consistent firing at all loads and speeds with load control by fuel regulation only. This results in an engine encompassing principles for high efficiency and wide fuel tolerance as well as low emissions.

According to the present invention, an internal combustion engine of the kind having a two-part combustion chamber including a primary chamber which is provided with fuel injection means and ignition means and a secondary chamber bounded by the piston, and wherein the secondary chamber communicates with the primary chamber through a throat which is positioned and dimensioned relative to the primary and secondary chambers such that, during a compression stroke of the piston in the secondary chamber, the air entering the primary chamber is caused to swirl therein a plurality of times is characterised in that the period over which fuel is injected into the primary chamber during a compression stroke corresponds, at full load operation, at most to about four times the period of a swirl and preferably to about once the period of a swirl, and corresponds at less than full load operation to a shorter period of swirl and preferably to less than the period of one swirl.

Preferably the quantity of air drawn in during the preceding induction stroke remains substantially unmodified over the complete load range.

Preferably also the peak swirl ratio occurring during a compression stroke lies within the range four to thirty inclusive, and most preferably is about nine.

The primary chamber preferably has a volume which is 65—85% of the total volume of the combustion chamber, and advantageously is about 75%.

It is desirable that injection of the fuel shall be by way of a soft spray nozzle, with an even distribution of fuel, wide spray angle and with a relatively low penetration and relatively low nozzle opening pressure compared with compression ignition engine nozzles of the single hole or pintle type.

Further, the sparking plug gap is desirably located just upstream of the throat, at a region between 20° and 100° and preferably about 60°, and for example disposed symmetrically between the side walls of the primary chamber, and advantageously projecting by between 0—50% of the radius of the primary chamber towards the centre thereof, and preferably about 20%.

In order that the nature of the invention may be readily ascertained, an embodiment of engine constructed in accordance therewith is hereinafter particularly described with reference to the accompanying drawings, wherein:—

Fig. 1 is a schematic axial section of the upper part of a cylinder block and cylinder head of an internal combustion engine.

Fig. 2 shows alternative compression ratio and throat width structures;

Figs. 3, 4, 5, and 6 are graphs to indicate experimental results from the engine of this embodiment.

In the drawings, only those parts of the engine are shown which are essential for the understanding of the improvement of the present invention, the remainder of the construction not being critical to the invention and being within the ambit of the man skilled in this art.

Referring to the drawing, Fig. 1, a cylinder block 1 has a cylinder bore 2 which receives a piston 3 having in its crown a recess 4 which forms part of the combustion chamber. In the cylinder head 5 there is an air inlet port 6 which can be closed and opened by a conventional inlet valve 7. In the cylinder head 5 there is provided a primary chamber 8 which communicates through a throat 9 with the cylinder bore 2. A conventional sparking plug 10 projects into the primary chamber 8 at a lower part thereof adjacent to the throat 9 and on the upstream side. A fuel injector 11 projects into the upper part of the primary chamber 8 upstream of the sparking plug.

The upper end wall 8*a* of the primary chamber is shown in a position for a low compression ratio. The wall 8*a* could be lowered towards throat 9 for achieving a higher compression ratio, see Fig. 2.

Throat 9 has been shown of a width corresponding to a throat to bore cross-sectional area ratio of 5.53% and could be made of greater or lesser width, see Fig. 2.

The combustion system involves mechanisms fundamental to the production of an exhaust relatively free of carbon monoxide, unburned hydrocarbons and oxides of nitrogen, through the employment of two distinct phases of burning.

The combustion chambe · as a whole consists basically of two regions, viz. a primary chamber and a secondary chamber constituted by the recess 4 (the piston being at top dead centre). The primary chamber 8, which comprises 65—85% of the clearance volume, is separated from the secondary chamber and swept volume of the engine cylinder bore 2 by the throat 9. The fuel injector 11 supplies fuel to the primary chamber 8 only. System geometry and injection timing are so arranged that at no time is raw fuel allowed to enter the secondary chamber which is bounded by the piston crown 3.

The engine operates on a four-stroke cycle. Air is drawn into the engine cylinder bore 2 past the inlet valve 7 which is open during the suction stroke. During the subsequent compression stroke, with the inlet valve closed, the air is compressed and flow occurs through the throat 9 from the cylinder bore 2 into the primary chamber 8. Fuel is injected by the injector 11 into the primary chamber 8 towards the end of the compression stroke. This mixes with the swirling air to form a combustible mixture which is carried downstream to the sparking plug 10 located just upstream of the throat 9 where combustion is initiated such that the flame kernel is passing the throat mouth once fully established. As the resulting flame propagates into the primary chamber 8 away from the region of the throat 9, the burning gas is rapidly forced into the relatively cool air contained in the secondary chamber. The consequent mixing and cooling of combustion products with the air contained in the secondary chamber permits further oxidation of combustion products without excessive formation of oxides of nitrogen.

The engine is designed to run with unthrottled air intake throughout its load range. To achieve this, the cylinder head incorporates the tangential throat 9 to the primary chamber 8 to establish an ordered air swirl on the compression stroke, at a peak swirl ratio, of say, 9:1. At full load the injection period corresponds approximately to one revolution of air swirl in the primary chamber 8 so that all the air in the primary chamber receives a slightly rich quantity of fuel. At light loads the injection period is shorter and corresponds to less than one revolution of air swirl and only part of the air in the primary chamber 8 receives fuel and timings are such that at minimum load the middle of the mixture cloud is passing the sparking plug at the optimum time for the spark.

By careful coordination of air motion, fuel spray characteristics, injection timing and duration, ignition timing and nozzle and sparking plug location it is possible to ensure consistent ignition at all loads and speeds, and the engine is thus operated throughout its load range by fuel regulation only.

The combustion chamber differs from a swirl chamber compression ignition engine in geometry in that a larger proportion of the clearance volume occupies the primary chamber, the compression ratio is lower and the throat is larger to give considerably lower rates of swirl. Moreover, with spark ignition it is essential to establish a finite volume of well-mixed reactants of approximately stoichiometric proportions around the sparking plug 10 at the optimum time for ignition so that, especially in its early stages, the combustion is governed by homogeneous processes rather than by mixing rates as with the diffusion or droplet burning characteristic of compression ignition engines.

The design of the throat 9 is important for several reasons:

a) The size and location of the throat affects compression-induced swirl and turbulence in the primary chamber 8. A coordination of swirl rate and injection rate provides optimum mixture preparation.

b) The compression-induced swirl and turbulence in the primary chamber 8 affect the rate of burning which controls the rate of pressure rise. Too high a rate of pressure rise results in combustion roughness.

c) The size of the throat 9 affects the rate of mixing of combustion products with secondary air and a high efflux rate from the throat 9 is desirable from this respect because of the time dependence of nitrogen oxides formation.

d) Heat transfer from the hot gases to the chamber walls rises rapidly with increase in velocity of the gases.

e) The ignition energy required to spark ignite the mixture cloud rises considerably with mixture velocity.

Optimization of throat size and location is desirable in that the resulting swirl rate must be high enough to ensure adequate distribution and mixing of fuel and air, but must not be too high to result in combustion roughness, high heat losses and unduly high ignition energy requirements.

Description of prototype engine

The prototype engine was a water cooled single cylinder unit of 105 mm bore and 130 mm stroke with pushrod operated valves.

The cylinder head was fabricated in mild steel and incorporated varying size top plugs to complete the top surface of the primary chamber so that the proportion of clearance volume occupied by the chamber could be altered. Two sizes were tested. As they altered the compression ratio of the engine too, they were known as the 9:1 and 11:1 N.C.R. (nominal compression ratio) plugs. The proportions of clearance volume occupied by the primary chamber (V_1/V_0) were respectively 72% and 64%. It also incorporated varying size inserts to alter

the width of the throat to give throat to cylinder bore cross-sectional area ratios (s/A) of 3.69%, 5.53% and 7.38%. Mathematical modelling at the design stage indicated that optimum coordination of primary chamber swirl rate and fuel injection rate occurred at a throat cross-sectional area ratio of around 5.5%.

The fuel injection system comprised a soft spray nozzle of the outwardly-opening poppet type with an even distribution of fuel, wide spray angle and with a low penetration and relatively low nozzle opening pressure compared with compression ignition engine nozzles of the single hole or pintle type. Rate of injection was relatively low.

The ignition system was a standard automotive coil/contact breaker system and a conventional sparking plug with good anti-fouling characteristics.

Performance of prototype engine

Operation was with wide open throttle unless otherwise stated, the fuel used was 97RON gasoline and the cooling water exit temperature was maintained at 79—82°C. Indicated m.e.p.s. were obtained by adding motoring m.e.p.s to the brake m.e.p.s. and subtracting 0.14 bar to take account of compression-expansion heat losses on motoring.

Basic configuration

Results are shown in Figs. 3 and 4 for the 9:1N.C.R. plug in the primary chamber and the medium throat width (s/A=5.53%).

1) The engine runs misfire-free throughout its load range and over the speed range tested with constant static injection timing.

2) From high load i.s.f.c. decreases steadily with increasing air-fuel ratio (A/F), in accordance with the theory of stratified charge combustion, until the overall mixture strength is weaker than about 0.4 stoichiometric, where further weakening causes i.s.f.c. to slightly rise. It is in this operating zone of very high overall A/F ratios where slow burning and homogeneous quenching effects are likely to show as mixture dispersion problems become more severe. The i.s.f.c. v A/F ratio characteristics are very flat with minima around 185g/kWh. For all A/F ratios weaker than 21:1 up to no load at 60:1, and for all the speeds tested, the i.s.f.c. lies between 185 and 240g/kWh. The A/F ratios for minimum i.s.f.c.s are very weak compared with conventional spark ignition engines.

3) Maximum i.m.e.p.s occur at overall mixture strengths somewhat weaker than stoichiometric, becoming progressively weaker as speed is decreased. Thus complete air utilisation is not achieved.

4) The engine operates with injection late on the compression stroke and is fairly insensitive to ignition timing. Mean best torque (M.b.t.) spark timing advances as air-fuel ratio increases, indicating that burning speeds are lower at weaker overall mixture strengths.

5) At all loads, right down to no load, more fuel is required at throttle openings other than wide-open in order to maintain constant speed. Thus throughout the load range, even down to idle, the engine is more economical at wide-open throttle and therefore with the weakest overall air-fuel ratio.

6) Exhaust gas temperatures are low compared with those of a conventional spark ignition engine (approximately 600°C for the latter regardless of load) and fall approximately linearly with fall in i.m.e.p. from 500°C at 8.0 bar to 330°C at 5.8 bar at 20.0 rev/s.

7) The optimum overall air-fuel ratio for cold-starting proves to be around 27:1.

8) Indicator diagrams exhibit a fairly moderate initial rate of pressure rise. A combustion knock is evident, but not harsh, at high loads. At light loads the engine is extremely quiet and smooth, almost like a steam engine. Cyclic dispersion is low.

Variation in throat width

Fig. 5 compares the performance of the engine with the narrow throat width (s/A=3.69%) with the medium throat width (s/A=5.53%). With the narrow throat, the engine performs poorer and showed itself unable to run at light loads, heat losses to the coolant are higher and combustion rougher.

With the wide throat, (s/A=7.38%) consistent firing was not obtained at all, indicating inadequate mixture motion in the primary chamber.

These tests demonstrate the importance in mixture preparation of optimising the coordination between air motion and fuel injection rate by means of throat size and orientation.

Variation in primary chamber size

Fig. 6 compares the performance of the engine with the 11:1 N.C.R. primary chamber top plug (V_1/V_0=64%) with the 9:1 N.C.R. primary chamber top plug (V_1/V_0=72%).

With the smaller primary chamber size, the engine gives a higher efficiency at all air-fuel ratios weaker than 23:1 with a minimum i.s.f.c. of 180g/kWh at an air-fuel ratio of 38:1. However maximum m.e.p. is lower and occurs at a higher overall air-fuel ratio showing that maximum air utilisation is lower. Cyclic dispersion is even smaller at the higher compression ratio.

However, the 9:1 N.C.R. results scaled up to compensate for the increased air cycle efficiency and higher volumetric efficiency (an improved design of inlet valve was fitted) give considerably higher i.m.e.p.s for the 9:1 N.C.R. configuration throughout the load range, the difference tapering off evenly to zero at no load. This is probably because as fuelling is increased the 11:1 N.C.R. configuration, with its lower proportion of clearance volume in the primary chamber, depends more on the secondary combustion process for fuel utilisation. This necessarily occurs later in the expansion stroke and therefore is not

exploited as effectively for the production of useful work.

Conclusions

1) It has been found possible to construct a spark ignition engine that will work throughout its load range at wide-open throttle by a coordination of air swirl and injection characteristics, to selectively distribute the fuel in a primary chamber occupying around 75% of the clearance volume.

2) The first practical embodiment of engine shows a low minimum i.s.f.c., 180g/kWh, very flat i.s.f.c. v air-fuel ratio characteristics, closely similar to a compression ignition engine's, a maximum i.m.e.p. of 8.6 bar and the ability to operate over a wide range of overall air-fuel ratios, 13:1 to 61:1.

At medium and light loads the i.s.f.c.s of the improved engine are thus markedly superior to those of a conventional spark ignition engine. Maximum m.e.p. is limited by air utilisation but this can be improved by detailed attention to the fuel injection equipment and combustion chamber design and advantage may be taken of a higher compression ratio and a higher volumetric efficiency than permitted in a conventional spark ignition engine. Combustion is smoother than with a compression ignition engine, especially at light loads, making the improved engine more acceptable for passenger vehicle use.

3) A comparison of the improved engine with a Newhall and El-Messiri engine constructed by modifying a C.F.R. engine shows that the improved engine has:

a) A much lower minimum i.s.f.c.

b) A much flatter i.s.f.c. v air-fuel ratio characteristic

c) A much higher maximum i.m.e.p.

d) The ability to run over a much wider range of overall air-fuel ratios.

4) With injection system parameters matched to the air swirl in the primary chamber the engine is very insensitive to injection timing and will run through its complete load range and over the speed range tested with fixed static injection timing. The engine operates with injection timings late on the compression stroke giving it the potential for wide fuel tolerance.

Ignition timing is desirably advanced as overall air-fuel ratio is weakened.

5) The relative sizes of the primary and secondary chambers affect efficiency and the effect of this is more pronounced at high load than at light load.

6) The system incorporates a two-stage combustion process that gives it the potential for low exhaust concentrations of oxides of nitrogen, hydrocarbons and carbon monoxide.

Claims

1. An internal combustion engine of the kind having a two-part combustion chamber including a primary chamber which is provided with fuel injection means and ignition means and a secondary chamber bounded by the piston, and wherein the secondary chamber communicates with the primary chamber through a throat which is positioned and dimensioned relative to the primary and secondary chambers such that during a compression stroke of the piston in the secondary chamber, the air entering the primary chamber is caused to swirl therein a plurality of times is characterised in that the period over which fuel is injected into the primary chamber during a compression stroke corresponds, at full load operation at most to about four times the period of a swirl and preferably to about once the period of a swirl, and corresponds at less than full load operation to a shorter period of swirl and preferably to less than the period of one swirl.

2. An internal combustion engine according to claim 1 wherein the quantity of air drawn in during the preceding induction stroke remains substantially unmodified over the complete load range.

3. An internal combustion engine according to claim 1 wherein the peak swirl ratio occurring during a compression stroke lies within the range four to thirty inclusive and most preferably is about nine.

4. An internal combustion engine according to claim 1 wherein the primary chamber has a volume which is 65—85% of the total volume of the combustion chamber and advantageously is about 75%.

5. An internal combustion engine according to claim 1 wherein injection of the fuel shall be by way of a soft spray nozzle with an even distribution of fuel, wide spray angle and with a relatively low penetration and relatively low nozzle opening pressure compared with compression ignition engine nozzles of the single hole or pintle type.

6. An internal combustion engine according to claim 1 wherein the sparking plug gap is located just upstream of the throat at a region between 20° and 100° and preferably about 60° and for example disposed symmetrically between the side walls of the primary chamber and projecting by between 0—50% of the radius of the primary chamber towards the centre thereof and preferably about 20%.

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